is always very great, the force may be taken as equal to - , where W lb.

S^r

is the weight of the valve or valves if two are used, *v* ft. per second is the velocity of the eccentric circle, and *r* ft. is the radius. This gives the total force due to the inertia of the valve. In addition, there is the friction of the valve itself and of the valve-rod packing, and, as these are indefinite quantities, it is usual to take twice the force found above and then proceed with regard to the stresses at the bottom of the thread in the valve rod and also the bolts, as in the case of connecting-rods. The area of the valve-rod pin may be decided in the same way, but the bearing pressure is usually kept lower than that on the crosshead pin or gudgeon, say 500 to 600 lb. per square inch. Generally the design of these parts is the result of experience and is greatly decided by their " look " and mutual proportion to each other

Crank-shafts. — It is almost impossible to compute the actual stresses on crank-shafts of multi- cylinder engines due to the combined effect of twisting and bending. The varying forces and bearing reactions are much too complex. In practice, formulas are applied which experience has shown to give good results.

A formula often used for the diameter of the crank-shafts of high-speed steam and similar engines is of the form $d^3 = C \times \frac{HP}{N}$, where C is a constant, but which varies for different types of engine. Now $\frac{HP}{N}$ is proportional to a torque, for, taking HP = $\frac{2\pi R F N}{33000}$, where F lb. is a force applied at radius R ft., and N is r.p.m., $\frac{HP}{N} = \frac{2\pi F R}{33000}$ and is thus proportional to the torque FR. The formula given above is seen to be of the right form, for the resistance of a circular shaft to both bending and twisting varies as the cube of the diameter. F X R is, of course, the mean torque due to the high pressure developed, but, owing to the variations in the turning effort on the crank-pin, the maximum torque is greater. In compound

engines it is 1-8 times the mean, and in triples it is 1-4 times.

The constant C, however, allows for all these considerations, and varies from 150 to 170 for compound engines with opposite cranks to 130 to 150 for triple-expansion engines with three cranks placed at 120° to each other.

It is usual to make the crank-shaft of uniform diameter throughout, with the exception that the bearing nearest the fly-wheel is made about 10 per cent larger in diameter, to allow for the stress induced by the over-hanging fly-wheel and that portion of the weight of the armature or other heavy part which may be taken by the outer engine bearing. For the same reason the outer bearing is increased in length also.

The crank-pins are usually of the same diameter as the body of the shaft, and the length is such as to keep the pressure due to the piston load not more than 450 Ib. per square inch.